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Performance of V-type Stirling-cycle refrigerator for different working fluids

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ABSTRACT

The thermodynamic analysis of a V-type Stirling-cycle Refrigerator (VSR) is performed for air, hydrogen and helium as the working fluid and the performance of the VSR is investigated. The V-type Stirling-cycle refrigerator consists of expansion and compression spaces, cooler, heater and regenerator, and it is assumed that the control volumes are subjected to a periodic mass flow. The basic equations of the VSR are derived for per unit crank angle, so time does not appear in the equations. A computer program is prepared in FORTRAN, and the basic equations are solved iteratively. The mass, temperature and density of working fluid in each control volume are calculated for different charge pressures, engine speeds, and for fixed heater and cooler surface temperatures. The work, instantaneous pressure and the COP of the VSR are calculated. The results are obtained for different working fluids, and given by diagrams.

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Diminution de la consommation d'électricité dans les systèmes frigorifiques à plusieurs compresseurs

Mots clés : Système frigorifique ; Système à compression ; Compresseur à vis ; Modélisation ; Optimisation ; Régulation ; Vitesse variable ; Économie d'énergie

1. Introduction

The first Stirling-cycle machine was built by R. Stirling in 1816 as an alternative to the steam engine. The classical analysis of Stirling-cycle machine is given by Schmidt, and in the literature

considerable efforts have been made to develop and improve this analysis using different working fluids (Walker, 1973).

Tew et al. (1978) analyzed the thermodynamic characteristics of the Stirling-cycle engine, and simulated by software. The model represents the working space by a series of

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subdivisions, which is called the nodal model. Urieli and Kushnir (1982) shown that their analysis can be utilized in order to evaluate the various practical effects of non-ideal regenerators, heat exchangers, including heat transfer and pressure losses.

Discussing the irreversibilities Berchowitz (1992) optimized a free-piston Stirling-cycle refrigerator with reference to designing machines for operation at intermediate temperatures. Walker et al. (1992) gave the brief review of the previous work for the Stirling-cycle refrigerators.

Angelino and Invernizzi (1996) showed that the heat pumps based on Stirling cycle are positively influenced by real gas effects. Kaushik and Kumar (2000) presented an investigation of a finite-time thermodynamic analysis of an endoreversible Stirling engine to maximize the power output and the corresponding thermal efficiency of an endoreversible Stirling heat engine with heat loss in the regenerator.

Otaka et al. (2002) developed a prototype of β -type Stirling-cycle refrigerator using helium, hydrogen and nitrogen as a working fluid. A Stirling-cycle machine of 100-W capacity was designed and tested. The effect of the dead space and phase angle on the refrigeration capacity was investigated.

Heidrich et al. (2005) developed a mathematical model for numerical simulations of free-piston Stirling coolers. As an alternative for conventional vapor compression cycles, a Stirling cooler for domestic refrigerators is developed. The model explores the working gas thermodynamic behavior and evaluates the performance of the Stirling cooler components.

There are a few studies about V-type Stirling-cycle Refrigerator in the literature. The V-type Stirling-cycle Refrigerator (VSR) is analyzed by Ataer and Karabulut (2005). In this study the Stirling-cycle refrigerator was divided into 14 fixed control volumes which were subjected to a periodic mass flow. The work, instantaneous pressure and COP of the Stirling-cycle refrigerator are calculated and the results are given by diagrams. Le'an et al. (2008) designed and tested the V-type Stirling refrigerator. The power consumption and the coefficient of performance (COP) are investigated under various rotating speeds and charged pressures.

In this study it is aimed to design a new V-type Stirling Refrigerator which has higher cooling capacity and the COP as a new alternative for the vapor compression-type refrigerator.

In this study the control volume analysis of the VSR is performed to determine the COP and cooling load of the VSR for the different working parameter. The refrigerator is divided into the control volumes. Heat is transferred to the refrigerator in the expansion space and heater, and transferred to the environment in the compression space and cooler. The VSR consists of a cylinder with a piston on each side of the regenerator as shown on Fig. 1 and the heater and cooler consist of multiple passes. For the storage of heat copper wire mesh is used in the regenerator. As shown in Fig. 1, especially geometry of heater and cooler is designed to increase to the heat transfer.

Each control volume of the VSR is exposed to a periodic mass flow. The control volumes of compression and expansion spaces are variable as regards to the motion of the pistons and the other volumes are fixed. The conservation of mass, momentum and energy equations are written for each control

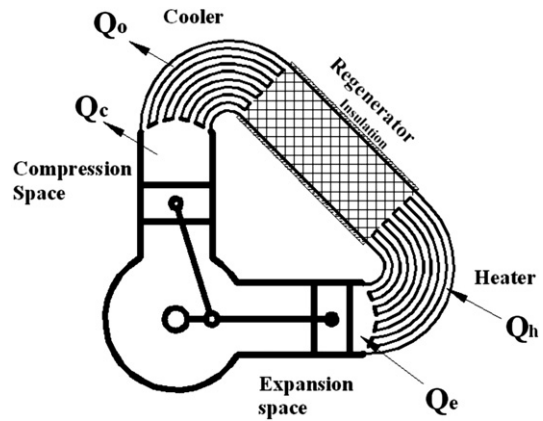


Fig. 1 – Schematic diagram of the VSR.

volume of the VSR, a computer program is written in FORTRAN, and the equations are solved iteratively.

2. Theory

In order to simplify the analysis the following assumptions are made:

- There is no leakage and the total mass of the working fluid is constant within the VSR.
- The working gas is treated as an ideal gas within the working conditions.
- The angular speed of the VSR is constant.
- The cyclic conditions are established in the VSR
- The gas flow in the VSR is one dimensional.

The variable volumes of the expansion and compression spaces of the VSR are expressed respectively

$$V_e = V_{e,d} + \frac{1}{2}V_{e,sr}(1 + \cos\theta) \quad (1)$$

and

$$V_c = V_{c,d} + \frac{1}{2}V_{c,sr}[1 + \cos(\theta - \phi)] \quad (2)$$

where $V_{e,d}$ and $V_{c,d}$ are the dead volumes of the expansion and compression volumes respectively, ϕ is the phase angle of the compression space relative to the expansion space. The mass balance for a control volume is written as

$$\frac{dM}{d\theta} = m_{in,i}^\theta - m_{out,i}^\theta \quad (3)$$

where $m_{in,i}^\theta$ and $m_{out,i}^\theta$ are the mass flow per unit crank angle of control volume “i”. The total mass of the VSR is constant and equal to

$$M_t = \sum_{i=1}^n M_i^\theta \quad (4)$$

where n is the number of the nodes. The flow in the control volumes of the VSR is compressible flow, and the simplified momentum equation is written as

$$\frac{\partial P}{\partial x} = -\frac{f}{D_{hy}} \frac{1}{2} \rho U^2 \quad (5)$$

The energy balance equation for a control volume is written as

$$Q_\theta - W_\theta = \frac{dE}{d\theta} \quad (6)$$

Eq. (6) can also be written as

$$\frac{dU}{d\theta} = \frac{dQ}{d\theta} + \frac{dM}{d\theta}(h_{in} - h_{out}) - \frac{dW}{d\theta} \quad (7)$$

The heat transfer per unit crank angle in Eq. (7) is defined as

$$\frac{dQ}{d\theta} = h_\theta A (T_w - T) \quad (8)$$

where h_θ is called the angular heat transfer coefficient. The work done per unit crank angle in Eq. (7) is written as

$$\frac{dW}{d\theta} = -P \frac{dV}{d\theta} \quad (9)$$

The variation of internal energy per unit crank angle is expressed as

$$\frac{dU}{d\theta} = MC_v \frac{dT}{d\theta} + TC_v \frac{dM}{d\theta} \quad (10)$$

Substituting Eqs. (8)–(10) into Eq. (6) one can obtain

$$MC_v \frac{dT}{d\theta} + TC_v \frac{dM}{d\theta} = h_\theta A (T_w - T) + (m_{in} C_p T_{in} - m_{out} C_p T_{out}) - P \frac{dV}{d\theta} \quad (11)$$

Using the ideal gas equation and rearranging Eq. (11), one can obtain:

$$\frac{dT}{d\theta} = \frac{h_\theta A}{MC_p} (T_w - T) + \left(\frac{m_{in}}{M} T_{in} - \frac{m_{out}}{M} T_{out} \right) - \frac{T}{M} \frac{dM}{d\theta} + \frac{V}{MC_p} \frac{dP}{d\theta} \quad (12)$$

In the analysis the variation of temperatures of the control volumes with crank angle are calculated. The variation of temperature with crank angle given in Eq. (12) has three components:

$$\left. \frac{dT}{d\theta} \right|_t = \left. \frac{dT}{d\theta} \right|_{\text{Heat transfer}} + \left. \frac{dT}{d\theta} \right|_{\text{Flow}} + \left. \frac{dT}{d\theta} \right|_{\text{Pressure}} \quad (13)$$

These three components can be written as

$$\left. \frac{dT}{d\theta} \right|_{\text{Heat transfer}} = \frac{h_\theta A}{MC_p} (T_w - T) \quad (14)$$

$$\left. \frac{dT}{d\theta} \right|_{\text{Flow}} = \left(\frac{m_{in}}{M} T_{in} - \frac{m_{out}}{M} T_{out} \right) - \frac{T}{M} \frac{dM}{d\theta} \quad (15)$$

and

$$\left. \frac{dT}{d\theta} \right|_{\text{Pressure}} = \frac{V}{MC_p} \frac{dP}{d\theta} \quad (16)$$

3. Numerical method

Using the mass balance equation given by Eq. (3), the inlet and outlet mass for i th control volume per unit crank angle in numerical form are written respectively,

$$\Delta M_{in,i}^\theta = (M_{i-1}^{\theta-1} + M_{i-1}^{\theta-1} + \dots + M_1^{\theta-1}) - (M_i^\theta + M_{i-1}^\theta + \dots + M_1^\theta) \quad (17)$$

and

$$\Delta M_{out,i}^\theta = (M_{i+1}^\theta + M_{i+2}^\theta + \dots + M_n^\theta) - (M_{i+1}^{\theta-1} + M_{i+2}^{\theta-1} + \dots + M_n^{\theta-1}) \quad (18)$$

the pressure difference given by Eq. (5) is written as

$$\Delta P_{i,i-1}^\theta = -\frac{f_\theta}{2D_{h,i}} \rho_i^\theta \left(\frac{u_{in,i}^\theta}{\beta} \right)^2 \Delta x_{i,i-1} \quad (19)$$

where $u_{in,i}$ is the velocity of the gas at the inlet of the control volume, $D_{h,i}$ is the hydraulic diameter and Δx is the distance between two control volumes.

The total temperature the variation per unit crank angle by Eq. (13) is written as

$$\Delta T_i^\theta = \frac{h_{\theta,i} A_i^\theta}{M_i^{\theta-1} C_{p,i}^\theta} (T_{w,i}^{\theta-1} - T_i^{\theta*}) \Delta \theta + \left(\frac{m_{in,i}^\theta T_{in,i}^\theta}{M_i^{\theta-1}} - \frac{m_{out,i}^\theta T_{out,i}^\theta}{M_i^{\theta-1}} \right) \Delta \theta - \frac{(M_i^\theta - M_i^{\theta-1}) T_i^{\theta*}}{M_i^{\theta-1}} + \frac{V_i (P_i^\theta - P_i^{\theta-1})}{M_i^{\theta-1} C_{p,i}^\theta} \quad (20)$$

The temperature of the node “i” at a crank angle θ can be written as

$$T_i^\theta = T_i^{\theta-1} + \Delta T_i^\theta \quad (21)$$

where ΔT_i^θ is given by Eq. (20). The following initial and boundary conditions are used to calculate T_i^θ . The initial working pressure is equal to the charge pressure and temperature;

$$T = 300 \text{ K} \quad P = P_{ch} \quad \theta = 0^\circ \quad (22)$$

By assuming that there are constant surface temperatures for heater, cooler, compression and expansion space and adiabatic conditions for the regenerator, the numerical solution is obtained. On the Other hand initially it is assumed that the regenerator temperature varies linearly. For the solution with the constant surface temperatures of the compression space, cooler and expansion space, heater are taken as

$$T_{w,o} = T_{w,c} = 320 \text{ K} \quad \text{and} \quad T_{w,h} = T_{w,e} = 260 \text{ K} \quad (23)$$

The energy balance equation for the metal matrix is written as

$$M_{m,i} C_m (T_{w,i}^\theta - T_{w,i}^{\theta-1}) = h_{\theta,i} A_i (T_{w,i}^{\theta-1} - T_i^\theta) \Delta \theta \quad (24)$$

where $M_{m,i}$ is the mass of the matrix in the control volume and

Table 1 – Data used in the analysis of the VSR.

Engine speed	1000	rpm
Phase angle between of the pistons	90	degree
Porosity of regenerator	0.7	
Charge pressure	150	kPa
Compression volume	1130	cm ³
Expansion volume	2010	cm ³
Length of the cooler	0.12	m
Length of the heater	0.16	m

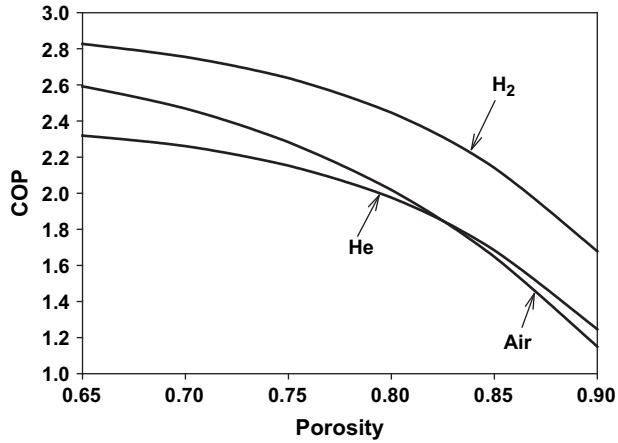


Fig. 2 – The variation of the COP of the VSR with the porosity of the regenerator for different working fluids.

C_m is the specific heat of the matrix. Using Eq. (24) the temperature of the matrix is written as

$$T_{w,i}^\theta = T_{w,i}^{\theta-1} + \frac{h_{\theta,i}^\theta A_i (T_{w,i}^{\theta-1} - T_i^\theta) \Delta\theta}{M_{m,i} C_m} \quad (25)$$

The total temperature change of matrix of the regenerator during a cycle is written as

$$\Delta T_w^k = \sum_{\theta=1}^{720} \left(\sum_{i=11}^{20} \frac{h_{\theta,i}^\theta A_i (T_{w,i}^\theta - T_i^\theta) \Delta\theta}{M_{m,i} C_m} \right) \quad (26)$$

where k is the number of cycle. Eq. (26) is used to correct the previous metal matrix temperature after a cycle, and metal matrix temperature is written as

$$T_{w,i}^k = T_{w,i}^{k-1} + \frac{\Delta T_w^k}{n_r} \quad (27)$$

The cooling load of the VSR and is written as

$$Q_c^\theta = \sum_{i=n_o}^n h_{\theta,i}^\theta A_i (T_{w,i}^\theta - T_i^\theta) \Delta\theta \quad (28)$$

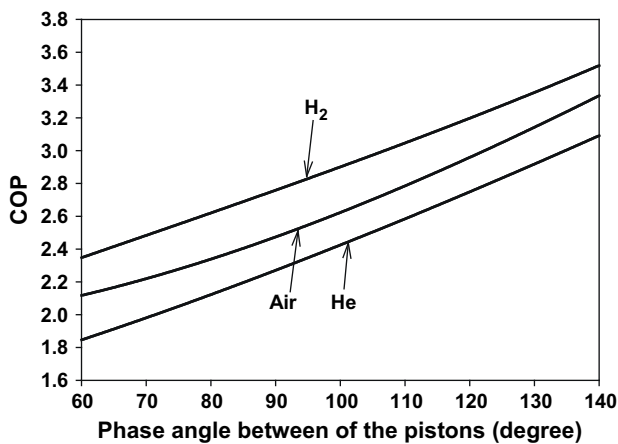


Fig. 3 – The variation of the COP with the phase angle for different working fluids.

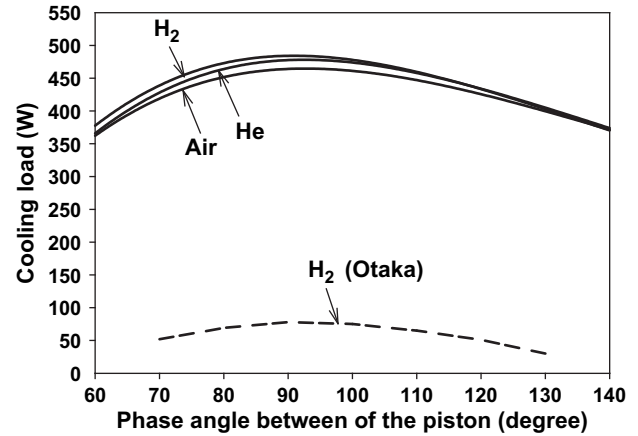


Fig. 4 – The variation of the cooling load of the VSR with the phase angle for different working fluids.

where n_o is the first control volume of the heater. The total work of the VSR is written as

$$W_o = \sum_{i=1}^{n_s} P_i^\theta (V_i^\theta - V_i^{\theta-1}) + \sum_{i=n_t}^n P_i^\theta (V_i^\theta - V_i^{\theta-1}) \quad (29)$$

where n_t is the first control volume when the working fluid enters to the expansion volume from the heater. The COP of the VSR is defined as

$$\text{COP} = Q_c / W_t \quad (30)$$

4. Method of solution

The basic equations of the VSR are obtained for per unit crank angle, so time does not appear in the equations. For the numerical analysis of the VSR a computer program is written in FORTRAN and the variation of the mass, pressure and temperature of the working fluid per unit crank angle and for each control volume are calculated. After each cycle the

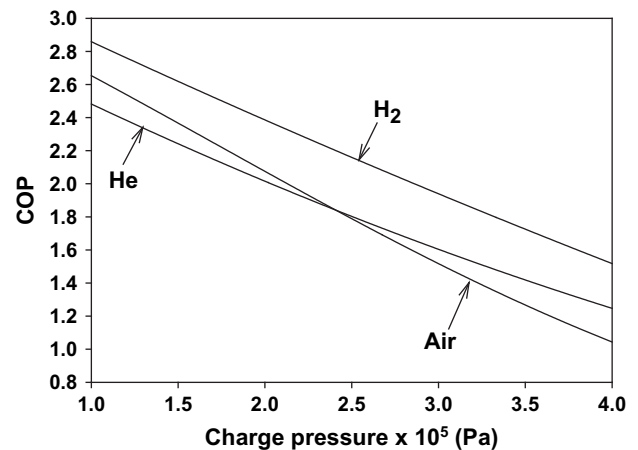


Fig. 5 – The variation of the COP of the VSR with charge pressure for different working fluids.

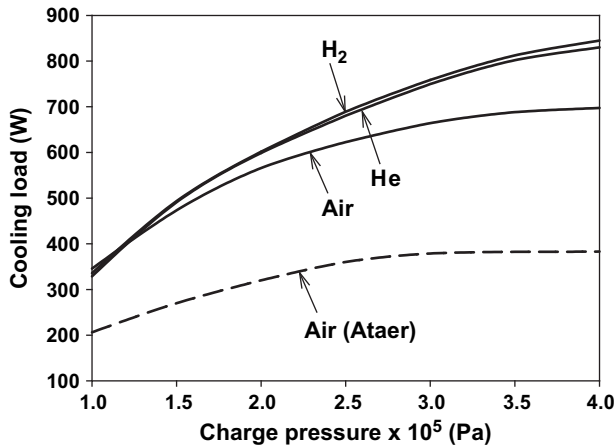


Fig. 6 – The variation of the cooling load with charge pressure for different working fluids.

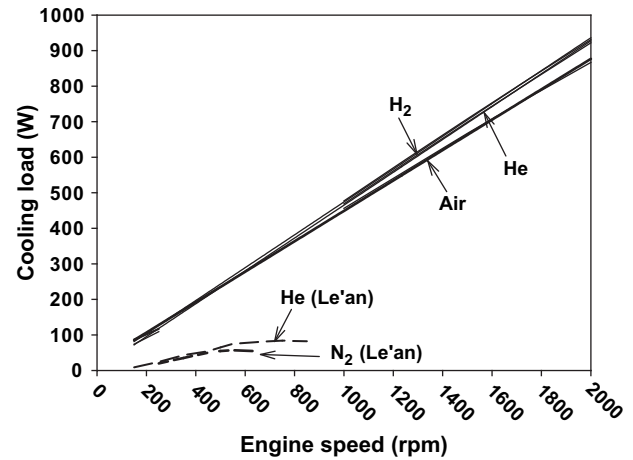


Fig. 8 – The variation of the cooling load with engine speed of the VSR for different working fluids.

cooling load, COP and work of the VSR are calculated. The method of analysis is given below:

1. Step: By means of FORTRAN program the volume, heat transfer areas, cross sectional areas and dimensions of the components of the VSR and amount of working fluid are calculated.
2. Step: The instantaneous pressure, the mass of fluid and the mass change per unit control volume of the VSR is calculated.
3. Step: The variation of the properties of the fluids with temperature is considered in the analysis.
4. Step: The velocities of the working fluids in meter per unit crank angle at the entry and exit of the control volumes are calculated.
5. Step: By considering the axial pressure losses new pressures of the control volumes are calculated.
6. Step: The heat transfer coefficients of the control volumes are calculated.

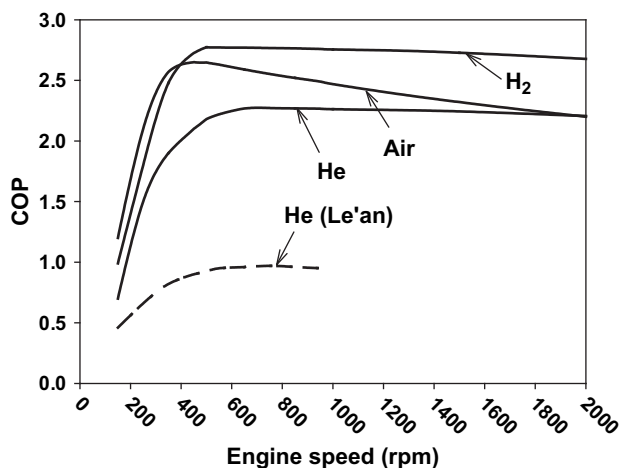


Fig. 7 – The variation of the COP with engine speed of the VSR for different working fluids.

7. Step: The temperatures of the control volumes after a θ crank angle are calculated iteratively. This is done using temperature change of each control volume, ΔT_i . Iteration is carried on when the following inequality for each control volume is satisfied:

$$|T_{ij+1}^{\theta} - T_{ij}^{\theta}| < 0,001 \text{ K} \quad (31)$$

where the j indicates the number of iteration.

8. Step: The regenerator of the VSR is assumed adiabatic in the analysis. The amount of heat transfer to the working fluid is equal to the amount of heat transfer to the metal matrix.
9. Step: The COP and the work done on the system per unit crank angle are calculated.
10. Step: The variation of the fluid properties with temperature is considered in the analysis.

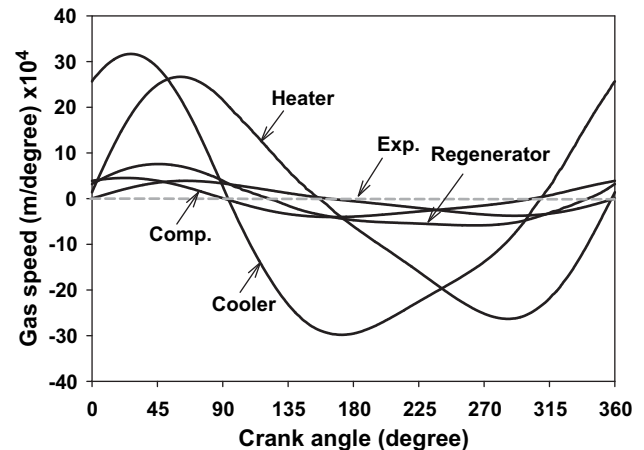


Fig. 9 – The variation of the working fluid velocity in the control volumes with the crank angle when air is used as working fluid.

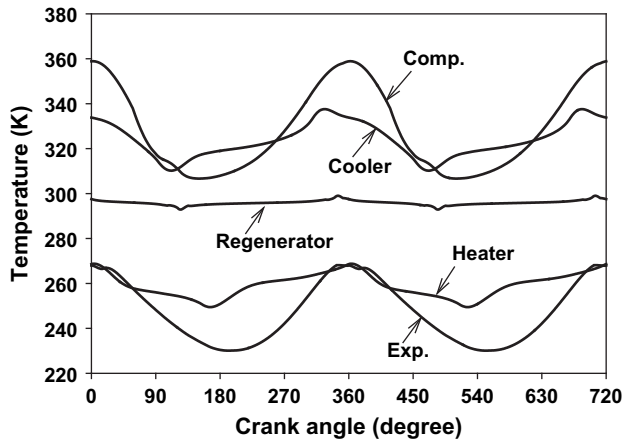


Fig. 10 – The variation of the temperatures of the working fluid with the crank angle when hydrogen is used.

11. Step: After a cycle the cooling load, COP and the work done on the VSR are calculated.

5. Results

The thermodynamic analysis of the VSR is performed using adiabatic regenerator boundary conditions. The basic equations of the VSR are solved numerically. A program is written in FORTRAN, and effects of different working fluids on the performance of the VSR are investigated. The results are obtained using the data given on Table 1, and given as diagrams.

The variation of COP of the VSR with the porosity of the regenerator is shown in Fig. 2 for the hydrogen, air and helium and for 90° phase angle. The porosity of the regenerator is taken between 0.65 and 0.90. As shown in the figure the COP of the VSR decreases with the porosity.

While the porosity of the regenerator decreases, heat transfer area of the matrix and the amount the heat transfer increases. The refrigerant goes into the heater at the low temperature. Thus the cooling load of the VSR increases. As an example; when the porosity, ϕ of the regenerator is 0.70, the COP of air, helium and hydrogen are 2.46, 2.26 and 2.77 respectively. When the porosity is 0.7, the cooling load of hydrogen, helium and air is 477 W, 468 W and 456 W, respectively.

The variation of the COP of the VSR with the phase angle between pistons is shown in Fig. 3. The COP of the VSR increases with the phase angle in the figure. This is due to the variation on the expansion and compression volumes. Fig. 4 shows the variation of the cooling load of the VSR with the phase angle. The results of the VSR and experimental results are taken from the literature for the β -type Stirling cooler and given in Fig. 4 for comparison when the hydrogen is used as the working fluid (Otaka et al., 2002). As it is seen, the cooling load increases with the phase angle and reaches a peak value at 90° and above 90° the cooling load of the VSR decreases. The maximum cooling load is obtained at the phase angle of 90° in both studies. The cooling load of the VSR is higher than the experimental study (Otaka et al., 2002). This is because of the ideal gas and no leakage assumption.

The variation of the COP of the VSR with charge pressure is shown in Fig. 5 for hydrogen, helium and air. As the charge pressure increases the COP of the VSR decreases in the figure. When the charge pressure increases, the work done on the system goes up. The fact that as the charge pressure increases, the cooling load of the VSR increases is shown in Fig. 6. Using hydrogen in the VSR has proved a better performance than air and helium. In the same figure, the result of the present study is higher than the result of the literature (Ataer and Karabulut (2005)). This is because of different geometry of the VSR.

The variation of the COP of the VSR with the engine speed for the hydrogen, helium and air is shown in Fig. 7 where the results are obtained for 150 kPa charge pressure. As the engine speed increases, there is a slight decrease on the COP of the VSR, but when the air is used, the slope of the decrease of the COP is high. The experimental results (Le'an et al., 2008) are also given on the same figure. The experimental results are considerably less than our results due to multiple passes in the heater and cooler.

The variation of the cooling load of the VSR with engine speed is given in Fig. 8 for 150 kPa charge pressure. As the engine speed increases the cooling load of the VSR increases. The increase in the cooling loads for three different working fluids is very close to each other. The cooling load of the VSR is almost the same and varies linearly for three working fluids. At the engine speed 1400 rpm, when hydrogen, helium and air are used for the working fluid, the cooling load of the VSR is 662, 653 and 624 W, respectively. The cooling load of the experimental study (Le'an et al., 2008) is lower than the results of this study.

Table 2 – Average temperature of working fluid in volumes for different working fluids.

Crank angle	Working fluids	Fluid temperature (K)				
		Compression volume	Cooler	Regenerator.	Heater	Expansion volume
0°	Air	366.57	334.30	298.19	268.61	269.61
	Helium	377.68	340.14	299.40	272.03	271.48
	Hydrogen	358.84	333.82	297.51	268.58	267.95
180°	Air	306.16	318.92	295.24	250.19	228.12
	Helium	302.91	318.75	296.79	247.50	219.24
	Hydrogen	307.93	319.01	295.35	250.85	230.41

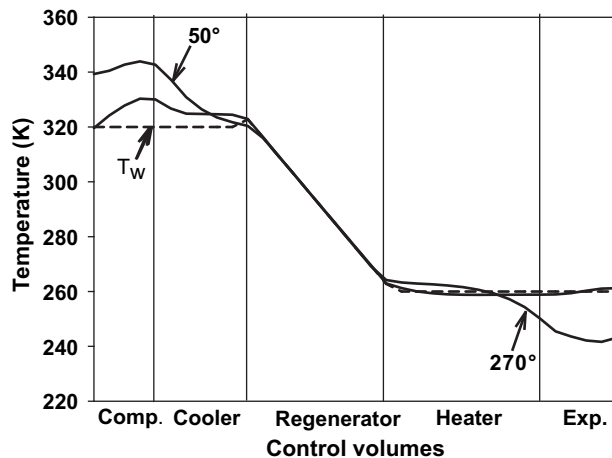


Fig. 11 – The variation of the working fluid temperature with control volumes when hydrogen is used as working fluid.

The variation of the fluid velocity with the crank angle for different control volumes of the VSR is shown in Fig. 9, when air is used as the working fluid. These velocities are given for the control volumes in the compression volume, cooler, regenerator, heater and expansion volume. As seen in the figure, the speed oscillation of working fluid is high in the heater and cooler.

The variation of the temperature in the control volumes at different units of the VSR with the crank angle is shown in Fig. 10 for hydrogen, and these are the temperatures of the compression volume, cooler, regenerator, heater and expansion volume. As seen in the figure, temperature oscillations of the compression and expansion volumes are high, and in the regenerator it is very low. As expected, while the fluid temperatures of the control volumes are decreasing during the expansion half-cycle, they are increasing during the compression half-cycle. At two different crank angles and for three working fluids the average temperature of the fluid in the compression volume, cooler, regenerator, heater and expansion volume are given in Table 2.

When hydrogen used as the working fluid the variation of the fluid temperature with the control volumes is shown in Fig. 11 for the crank angles of 50° and 270°. The metal temperature the variation in the regenerator is also given in the figure for comparison.

The copper or aluminum is used as the matrix of the regenerator. The type of metal matrix of the regenerator affects the COP and the cooling load of the VSR. When copper is used as the solid matrix in the regenerator, the COP of the VSR is 1.633 and the cooling load is 441 W, and when aluminum is used the COP and the cooling load of the VSR are 1.61 and 435 W respectively for hydrogen and for the charge pressure of 150 kPa.

6. Conclusions

In this study, the thermodynamic analysis of the VSR is performed, and the performance of the VSR is investigated for three different working fluids; the hydrogen, helium and air.

In the analysis, the basic equations are derived for unit crank angle, and the time is not seen in the basic equations. This approach simplifies the analysis of the VSR. The analysis is performed for constant surface temperatures of the compression and expansion spaces, and the adiabatic boundary conditions in the regenerator.

When the hydrogen is used as the working fluid, the COP of the VSR is higher than the VSR that uses helium or air. This is due to low pressure drop and high specific heat of the hydrogen. The temperature distribution of the working fluid in the regenerator is approximately linear and this result is in agreement with the results given in the literature (Ataer and Karabulut, 2005). The results obtained for the regenerator matrix are made of copper and aluminum, and a better performance is obtained for the copper matrix.

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